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The Benefits of a Mid-Route Exhaust Gas Recirculation (EGR) System for Two-stage Boosted Engines

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Abstract

Exhaust Gas Recirculation (EGR) is a widely-known technique applied in internal combustion engines for controlling the combustion process and harmful emissions. The recirculation of gases can be achieved either by delivering burnt gases from upstream of the turbine to downstream of the compressor (short-route) or by taking the exhaust gas downstream the turbine and deliver upstream the compressor (long-route). Although long-route (LR) system is preferred for highly boosted engines due to the higher EGR availability at low engine speeds, it lacks a fast response time during transient performance compared to the short-route (SR) system.

This paper examines the potentials of introducing an alternative EGR route which can be applied in two-stage boosted engines. The proposed Mid-Route (MR) EGR system, **applied in a gasoline engine**, combines the benefits of the LR and SR **routes**. **The system** provides high EGR rates at all engine speeds while the transport delay in the case of transient operation is relatively short. The potential of a hybrid EGR system combining MR and LR **EGR** is examined and various components' (i.e. compressor, turbine and coolers) sizing and transient performance studies are performed to understand the trade-offs of the system.

It was demonstrated that MR could provide high EGR particularly at high and low engine speeds. A combination of MR and LR EGR can provide maximum EGR rates at all speeds **with a maximum fuel consumption penalty of 1.4% at engine speeds below 2,500rpm**. The reduction in EGR response time was of the magnitude of 50% while the faster EGR purging time combined with the smaller turbine implemented, dropped the load tip-in response time by 25%. The coolers' sizing study revealed that an LR EGR cooler is unnecessary whereas the MR EGR cooler can also be omitted when the flow is delivered prior an intercooler with a 25% larger cooling capacity than of the baseline engine.

Keywords

Gasoline, EGR, exhaust gas recirculation, short-route, mid-route, high-pressure EGR, low-pressure EGR, internal combustion engines

Introduction

Exhaust Gas Recirculation (EGR) is a widely-adopted technology in internal combustion engines for the reduction of NO_x formation and maintenance of the stringent emission regulations. The recirculation of exhaust gases has been proved not only to reduce the NO_x formation but at the same time reduce the fuel consumption and work as a more efficient suppressant of knock than excess air for gasoline engines¹⁻⁶.

The recirculation of exhaust gases back to the inlet manifold through an intercooler is a direct way to reduce the amount of available oxygen within the engine's cylinders. Also, the higher specific heat capacity of the EGR compared to the fresh air increases the heat capacity of the intake charge and therefore leads to lower in-cylinder peak flame temperatures and lower NO_x formation. The improvement in fuel consumption occurs due to the reduced pumping work, reduced heat losses to the cylinder walls and the reduction in the degree of dissociation in the high temperature burned gases⁷.

The implementation of the EGR technology is quite simple and straight-forward in naturally aspirated (NA) engines where the exhaust flow is diverted back to the inlet manifold as a result of the pressure difference existing between the exhaust and inlet sides of the engine. An additional back pressure valve is often implemented to ensure a positive pressure difference between the exhaust and intake sides and force the EGR to the intake system. The EGR flow is typically controlled by a throttling valve and is cooled down to increase the volumetric efficiency of the engine. However, in turbocharged engines, the recirculation of the exhaust flow back to the inlet manifold is a more challenging task due to the high inlet pressures. Different EGR architectures have been proposed and implemented over the last years and their main advantages and disadvantages are presented below.

Short route (SR) or High-Pressure (HP) EGR is nowadays the most common EGR architecture in internal combustion engines. In this architecture, the exhaust gas is recycled back to the inlet manifold from the upstream of the turbine to downstream of the compressor (or downstream of the intercooler if applicable). The benefits of this EGR system are the fast transient response and the fact that the compressor and the intercooler are not exposed to the corrosive EGR gases containing sulphur and nitrogen compounds and therefore fouling is prevented. However, the main drawback of this system is the high pre-turbine pressure (smaller turbine or a more closed wastegate position) required for the exhaust flow to circulate back to the inlet side which often leads to increased pumping losses. Also, the fact that part of the exhaust gas is diverted before the turbine reduces the exhaust mass flow through the component and could potentially limit the performance of the turbocharger.

Long Route (LR) or Low-Pressure (LP) EGR system is an alternative EGR routing architecture which delivers exhaust gas from downstream of the turbine (or downstream of aftertreatment devices if applicable) to upstream of the compressor. The main drawbacks of this system are the long distance the EGR flow needs to cover, which restricts the transient response time, and the fouling of the compressor and intercooler. However, the LR EGR system benefits from lower intake charge temperatures which can increase the volumetric efficiency of the engine for a certain EGR rate. Low-pressure EGR also eliminates the need for high exhaust backpressure and therefore reduces the pumping losses. Additionally, engines fitted with a long EGR route benefit from a high EGR uniformity at the inlet manifold and subsequently an increased cylinder-to-cylinder EGR balance.

Extensive research has been previously conducted on the advantages and drawbacks of the two EGR systems on the performance and emission generation of internal combustion engines. In 2000, Lundqvist et al.⁸ equipped a Euro 0 heavy-duty diesel engine with a particulate trap and different EGR systems to meet the Euro IV emission levels with a minimum penalty on engine performance and fuel consumption. The authors demonstrated that the SR EGR system was more efficient than the LR EGR both regarding NO_x reduction and fuel consumption. Vitek et al.⁹ presented a detailed theoretical study of different EGR routes using a GT-power engine model. The authors considered four different EGR routes, an SR EGR, an LR EGR from downstream of the compressor, an LR EGR from downstream of the Diesel Particulate Filter (DPF) and a combined SR and LR EGR system. The authors highlighted that from a BSFC point of view, the best variant is the LR one. The main advantage of this layout is that it uses the relatively high pressure loss of the DPF to create a

sufficient drop via the EGR valve. It can be described as a convenient use of natural pressure losses. They concluded that the final selection of an appropriate EGR route depends on the particular engine's requirements and a compromise is required.

Van Aken et al.¹⁰ investigated the effects of SR and LR EGR on a two-stage turbocharged 12l heavy duty diesel engine. They found that LR EGR is favourable for low loads and SR EGR is favourable for high loads when achieving high EGR rates (NO_x 0.5 g/kWh) with this engine configuration. The filtered clean long route EGR showed no clear advantages in PM emission reduction when compared to the SR EGR at the same lambda. The authors also observed the effects of EGR routing on the cylinder-to-cylinder EGR distribution by measuring the exhaust gas temperature differences per cylinder. They concluded that cylinder-to-cylinder temperature variation was decreased when LR EGR was applied which is most likely a result of an improved EGR distribution per cylinder.

Beatrice et al.¹¹ examined the benefits of cooled and extra-cooled LR EGR on a light duty 4-cylinder diesel engine of two litres displacement. The results obtained suggest an improvement of both fuel consumption and emissions with LR EGR compared to the classical SR EGR. The improvements were more evident as the speed and load increased and were observed both in conventional diesel and PCCI modes. The extra-cooled LR EGR showed only marginal advantages over the cooled LR one.

Millo et al.^{12, 13} performed experimental and computational analysis of different EGR systems for a passenger car fitted with a common rail diesel engine. As reported, three different EGR layouts (a Long Route, a Short Route and a combination of the two) were applied under steady state and transient engine operating conditions. It was shown that the use of LR EGR led to an extremely effective reduction of NO_x emissions up to 15% over the first 60s of the Extra-Urban driving cycle (EUDC). The authors mentioned the need of using a Dual Loop EGR layout, combining both an LR and an SR system to provide SR behaviour during fast transient and allow lower temperatures (and thus lower NO_x emissions) during quasi-steady-state operating conditions.

Reifarth and Ångström¹⁴ performed a 1D simulation for a 1.9l passenger car diesel engine equipped with a set of different ways of supplying EGR and a variable geometry turbine. In this work, three different EGR systems were employed. The hybrid system was a combination of LR and SR EGR. The second system was an LR EGR where a pump was used to drive the EGR-flow. This system allowed having an unthrottled exhaust flow, reducing the backpressure behind the turbine. The third system was an SR EGR with a reed valve. The transient analysis comparison showed advantages and disadvantages of both SR and LR systems. According to Reifarth and Ångström¹⁴, the time delay of the LR EGR is not a problem for positive load transients as this is compensated for with improved turbocharger performance compared to SR circuits. However, for negative load transients and thus positive EGR transients, the LR EGR response delay could lead to NO_x peaks during the first cycles of the load change. The authors also showed that a hybrid EGR approach could combine the advantages of both systems by implementing SR EGR at decreasing engine load conditions for a quick response and LR EGR at steady load points for increased efficiency. The implementation of an electrically driven pump in the LR EGR system was found to enhance the efficiency of the engine further, though leading to increased system cost.

Bermúdez et al.¹⁵ conducted a study of regulated and unregulated gaseous emissions in a light-duty diesel engine during the New European Driving Cycle (NEDC) using SR and LR EGR with different engine calibrations. The main conclusions of the study are that the fuel consumption increased with the use of LR EGR without engine calibration modifications due to the increase in the pumping losses and the lower engine efficiency. However, a reduction of 8.5% in total fuel consumption can be achieved by optimizing the backpressure valve and the start of injection. A 5% reduction of NO_x emission was shown for LR EGR without

engine calibration and the reduction was increased to 21% by changing the engine calibration. The major drawback of this was that carbon monoxide was three times higher and total hydrocarbons emissions were increased by 1.7 times when low-pressure EGR system was used due to the deceleration of the reaction rates of the mixtures and the lower combustion temperatures.

Heuwerker et al.^{16, 17} combined LR and SR EGR systems on a General Motors Z19DTH 1.9/ Euro 4 specification diesel engine equipped with a variable geometry turbine (VGT) to achieve low emissions and fuel consumption without sacrificing transient response capability due to the long EGR path. The authors concluded that hybrid EGR resulted in reductions in almost all recorded emissions for both steady state and transient operation. NO reductions of more than 50% were achieved under steady state and transient operation with a late injection strategy due to increased cooling and boost capacity provided by the LR EGR loop. However, a small increase of HC and soot under steady state conditions was shown when using a late injection combustion strategy.

Zamboni and Capobianco^{18, 19} studied the effects of SR and LR EGR systems on the consequences of in-cylinder pressure and heat release curves and the quantities mainly affecting the outset and development of combustion process and emissions such as intake pressure and mass flow rate. They proved that LR EGR circuit shows potential for enhanced NO_x reduction, limited interaction with the turbocharging system and wider opportunities to develop integrated control strategies for the main engine subsystems. Zamboni et al.²⁰ also demonstrated the benefits of managing LR and MR EGR systems and the variable nozzle turbine, aiming at major reductions in NO_x emissions while enhancing the fuel consumption of a Euro 5 DI diesel engine. They found that in suitable operating modes of tested conditions, NO_x emissions were reduced up to 58–66%. The brake specific fuel consumption decreased around 5–9.5% at low speed/load and 1.7–3.3% at medium loads. The activation of low-pressure loop confirmed to be beneficial for the turbocharger's performance, rising its rotational speed thus granting for a better transient response.

Roth et al.²¹ explored the PMEP benefits of mixing low and high-pressure EGR in the mid to upper speed and load ranges of turbo GDI engines. The authors established that cooled low-pressure EGR needs to be the primary path at speeds and loads above the knock limit of the engine. For a low load operation, where no cooling is required to minimize knock tendency, the high temperature of the uncooled high-pressure EGR helps to thermally de-throttle the engine and improve transient EGR control. At moderate speeds and loads and above, mixing cooled or uncooled high-pressure with low-pressure EGR was found to provide a small PMEP improvement with a BSFC benefit of less than 1%.

Suresh et al.²² focused on the challenges involved in selecting a single stage variable geometry turbocharger capable of delivering a 75hp/liter power density in addition to meeting light duty drive cycle emission requirements. The authors explored the benefits of LR and SR EGR systems for this light-duty engine architecture in detail along with outlining the possible strategy of dual loop EGR to reduce overall fuel consumption. The LR EGR technology showed promising results with respect to achieving a significant reduction in both NO_x and fuel consumption. Dual loop strategy involving both SR and LR EGR loops was found to expand the scope for further optimization by reducing pumping losses resulting in lower fuel consumption.

In this paper, a detailed numerical analysis for the optimum EGR routing in a two-stage boosted direct injection gasoline engine is conducted. The study initially analyses the availability of long and short route EGR under the same pre-turbine pressures at full load conditions. A scaling exercise is performed for the SR EGR system for understanding the effects of turbine and compressor sizing on the availability of EGR and the performance of the engine. Next, a new mid-route (MR) EGR system that collects exhaust gases from

upstream of the turbine and delivers between the first and second boosting stage is introduced and its performance is compared with the LR and SR systems. Two new hybrid EGR systems combining SR or MR and LR EGR paths are tested under steady-state and transient conditions. Finally, a coolers' sizing analysis is performed for understanding the cooling capacity demands for the intercooler and EGR coolers of the system.

Computational Models

The present numerical study was performed in a 1D gas dynamic environment using GT-Power software tool. The model used for this study represents a heavily downsized 2.0l DISI engine featuring low-pressure or 'long route' (LR) EGR. **The non-predictive SI Wiebe combustion model was applied and calibrated based on experimental results at various engine conditions.** The UB200 build of the engine features a first-stage wastegated turbocharger (TC) combined with a high-pressure supercharger (SC) engagement clutch and a full bypass system as shown in Figure 1. **The engine has been operated at torques of over 500Nm and maximum brake power of 285kW without any knock issues, therefore an engine knocking model has not need incorporated in this study. The full load curve of the engine is presented in Figure 2.**

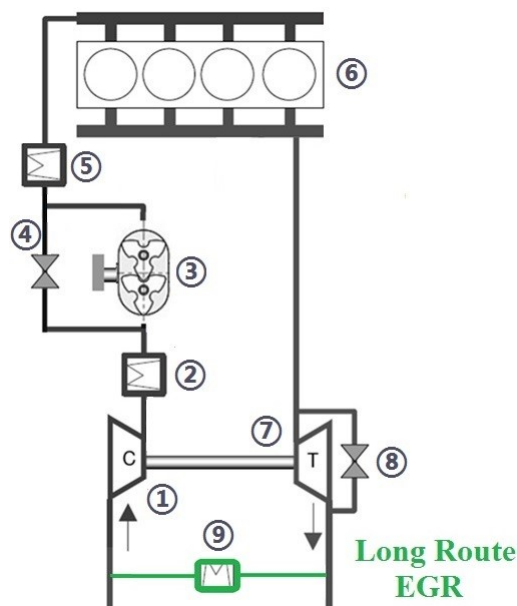


Figure 1. Schematic of the two-stage boosted DISI engine; ① Compressor, ② Intercooler, ③ Supercharger, ④ By-pass valve, ⑤ Aftercooler, ⑥ Engine block, ⑦ Turbine, ⑧ Wastegate, ⑨ LR EGR cooler.

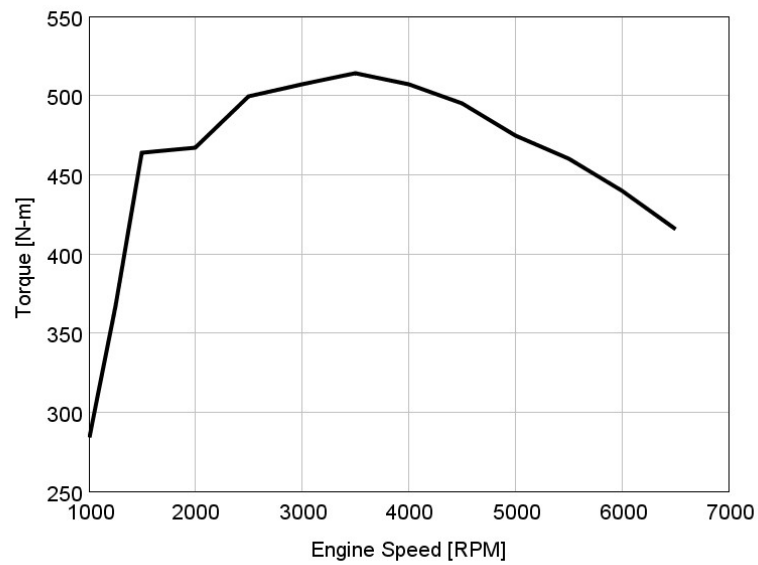


Figure 2. Limiting torque line of the Ultraboost UB200 engine.

The supercharger benefits of a drive pulley ratio of 5.9:1 and is only engaged at low to medium engine speeds, as shown in Figure 3, for minimizing torque dip. This layout has been preferred than the common supercharger-first system for taking advantage of the supercharger's charge-cooler even when it is clutched out. The engine design and control systems are described in detail in a publication by Turner et al.²³ and therefore only briefly summarized here. The engine model, previously used by Turner et al.^{23, 24}, has been developed and validated with experimental testing by Lotus Cars Limited as part of the 'Ultraboost' project, a collaborative research project which is co-funded by the Technology Strategy Board, the UK's innovation agency.

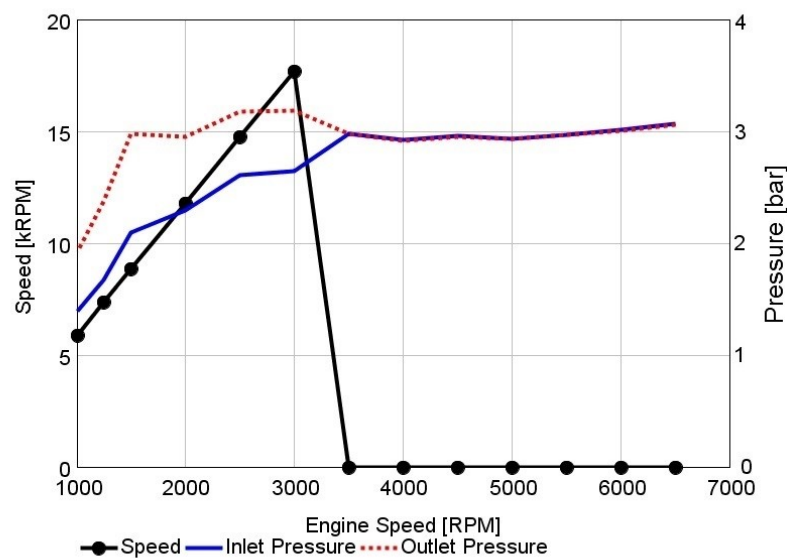


Figure 3. Supercharger's operating speeds and pressure levels at inlet and outlet in full load conditions (engaged for engine speeds of 3,000rpm and below).

The baseline and modified models used for this study are presented below.

Ultraboost model – Long route (LR) EGR (baseline model)

A ‘long route’ (LR) or low-pressure EGR path system is applied in the baseline engine. The LR path entails taking the post-turbine exhaust gases and introducing them upstream of the compressor at a location that facilitates good mixing given both the nature of the compressor and its distance from the intake plenum. The implementation of an EGR cooler is mandatory for improving engine swallowing capacity (volumetric efficiency) and take all the advantage of boosting systems.

Ultraboost model – Alternative EGR routings

The baseline model was modified to run with alternative EGR routes. As a first step, the model was modified to a ‘short route’ (SR) or high-pressure EGR path system by delivering exhaust gases from the exhaust manifold prior the turbine to the inlet manifold. This approach requires a high exhaust back pressure for allowing the exhaust flow to circulate back to the inlet side.

The second method was the implementation of a mid-route (MR) EGR by delivering the exhaust gases taken upstream of the turbine to the intake side downstream the compressor and prior the second stage boosting device. This approach can significantly increase the pressure difference between the exhaust and inlet sides at the points where the supercharger operates, reduce the pumping losses and increase the maximum possible EGR rate.

The distance that the EGR flow has to travel from the collection point to the engine’s cylinders for the three routes may include pipes, coolers and boosting devices and is analysed in Table 1.

Table 1. Piping length (including coolers) for the three EGR routes based on real data (LR) and logical suppositions for SR and MR systems.

EGR routing	Distance (m) (closed by-pass)	Distance (m) (open by-pass)	Comments
Short route	1.4	1.4	Actual distance
Medium route	2.4 + SC	2.5	Logical supposition
Long route	2.6 + TC + SC	2.7 + TC	Logical supposition

It is clear that the SR path, as expected, provides the shortest distance to be covered by the exhaust gases to reach the cylinders of the engine. On the other the hand, the difference between the mid- and long routes was considered to be very small. This is mainly due to the extra length needs to be covered for delivering the gas prior the aftercooler and the supercharger plus the extra length between the delivery point of the MR and the SR systems.

Methodology

The present investigation focuses on the availability of exhaust gas recirculation for the various systems under the same engine conditions without considering increasing the pre-turbine pressure which as shown in the literature could result in a deteriorated fuel efficiency. The simulations performed in this paper are divided into four main categories based on the type of study.

EGR availability

First, an EGR availability study was conducted for the three different routes. The targeted EGR rate for all models was 10% which is the maximum amount of EGR used at full load to improve the combustion knock-limit of the engine. The EGR rate is defined as,

$$\text{EGR rate} = \frac{\oint \dot{m}_{b,i} dt}{\oint (\dot{m}_{b,i} + \dot{m}_{ub,nf}) dt} * 100 \quad (1)$$

Where $(\dot{m}_{b,i})$ is the instantaneous mass flow rate of burned gas through all intake valves into cylinder i and $(\dot{m}_{ub,nf})$ is the instantaneous mass flow rate of unburned non-fuel gases past all intake valves.

The maximum amount of LR EGR depends on the backpressure generated as a result of the aftertreatment system of the engine. In most cases, the pressure difference between the exhaust and intake sides is high enough to achieve maximum EGR rates. On the other hand, SR and MR EGR systems rely on the pressure gradient from the exhaust manifold to the boosted area of the intake side. The high boost pressures often lead to small pressure gradients between the exhaust and intake sides and limited EGR rates.

Also, short route and mid-route EGR systems bypass the turbine and compressor as the gases are delivered from prior the turbine to the upstream of the compressor. Therefore, a smaller turbocharger can be implemented, meaning the potential for better transient response due to less inertia and possibly better fuel economy due to the reduced need for supercharger operation (in the case of an SR EGR system). A scaling sweep study was performed to assess the benefits of a smaller turbine for the SR and MR configurations compared to the baseline LR EGR system.

The scaling of the compressor and turbine was performed as suggested in the literature by Guillaume et al.²⁵. By restricting or enhancing the mass flow rate within a component (2), the size of the component can be simulated. The mass flow through the turbine and compressor was reduced from 10% up to 50% while the changes on the speed were considered using equation (3). **The scaling factors of the compressor and turbine were of the same magnitude for each case tested.**

$$\text{Mass flow multiplier} = \frac{\dot{m}^{baseline}}{\dot{m}^{scaled}} \quad (2)$$

$$\text{Speed multiplier} = \frac{1}{\sqrt{\text{Mass flow multiplier}}} \quad (3)$$

Where (\dot{m}) is the mass flow rate through the part (in the case of the turbine, the wastegate mass flow is excluded).

Hybrid EGR configurations

The results obtained in the EGR availability section suggest that both the SR and MR configurations cannot provide maximum EGR rates (10%) for all engine speeds tested. A hybrid approach was followed by combining short or mid-route EGR with the long route and the hybrid models were tested at all speeds for full load conditions. The LR supply works as a backup path and is only there to assist the MR or SR routes when targeted EGR rates cannot be achieved. The results were compared with the baseline model and an engine combustion analysis was performed.

Transient analysis

A comparison of the transient performance between the baseline model and the hybrid EGR systems was performed. Two transient studies were conducted including a load tip-in from 5 bar BMEP to full load with sudden closure of the EGR valve for faster performance and a 0 to 10% EGR 'tip-in' event at full load conditions. Both of the transient studies were performed at 1,500 and 3,500rpm representing cases with and without the supercharger activated.

Coolers sizing

The baseline model benefits of an LR EGR cooler, an intercooler after the compressor and an aftercooler placed post the supercharger. On the other hand, the hybrid models potentially require an extra cooler for the additional EGR path. **Each cooler is represented by a number of identical pipes with an imposing wall temperature slightly higher than the coolant's temperature. The heat transfer and friction multipliers were adjusted to meet the coolers' performance obtained from the experimental results.**

For the coolers sizing investigation, the aftercooler's cooling capacity was kept constant to provide the same cooling capabilities as in the baseline model. The studies performed and the approach followed for the baseline and hybrid models with a 10% scaled turbocharger are as listed.

- I. An investigation of the LR EGR cooler capacity for the hybrid models
 1. The intake manifold temperature was kept constant at 318K for all speeds by controlling the intercooler's and SR or MR EGR coolers' performance
 2. The pre- compressor's temperatures were targeted at the same levels as in the baseline engine
 3. The amount of heat dissipated by the LR EGR cooler was calculated for the hybrid cases and compared to the baseline
- II. An investigation on the cooling capacity of the intercooler without an LR EGR cooler
 1. The LR EGR cooler was replaced by a straight pipe
 2. The intake manifold temperature was kept constant at 318K for all speeds by controlling the intercooler's and SR or MR EGR cooler's performance
 3. The intercooler's intake charge temperature and any possible effects on the performance of the compressor were addressed
 4. The intercooler's cooling demands were calculated (**assuming post SR or MR EGR temperature equal to the IM requested temperature**)
- III. Intercooler's sizing study with various MR or SR EGR coolers
 1. The pre-compressor temperature was kept fixed as in the baseline model
 2. The cooling capacity of the SR or MR cooler was set at similar rates to the LR cooler of the baseline model and up to 50% higher
 3. The heat needs to be dissipated by the IC for achieving an intake temperature of 318K was calculated for various SR or MR EGR coolers
- IV. IC sizing for different MR options as shown in Figure 4
 1. For the MR case without an MR EGR cooler, the IC's performance was controlled to achieve an intake temperature of 318K
 2. For the MR case with an MR EGR cooler (but no LR EGR cooler), the performance of the MR EGR cooler was set to 20% higher than the baseline LR EGR cooler and the IC's performance was adjusted to achieve an intake temperature of 318K

All the coolers sizing cases studied are summarized in Table 2.

Table 2. Summary of the coolers sizing cases (? indicates the cooler under investigation for each case, X indicates a cooler not in use).

Case	LR cooler	MR or SR cooler	IC	AC
Case I	?	Maximum cooling capacity of the baseline LR cooler is provided	Controlled to meet targeted IM	Fixed
Case II	X	Maximum cooling capacity of the baseline LR cooler is provided	?	Fixed
Case III	Adjusted to meet targeted pre-compressor temperature	Maximum cooling capacity and up to 50% higher of the baseline LR cooler is provided	?	Fixed
Case IV 1	Adjusted to meet targeted pre-compressor temperature	X	?	Fixed
Case IV 2	X	Performance of the MR EGR cooler was set to 20% higher than the baseline LR EGR cooler	?	Fixed

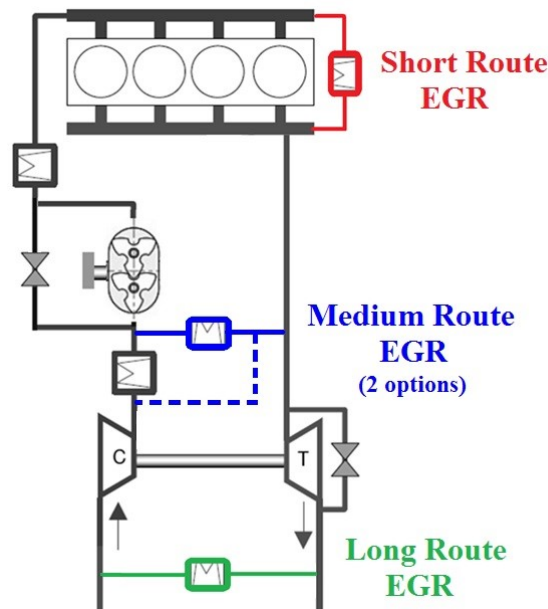


Figure 4. Schematic diagram of SR, LR and two different MR options with or without EGR cooler.

Results and discussion

EGR Availability

Long Route vs. Short Route

Long route EGR system has been previously proved preferable for reduced pumping losses and improved fuel economy in downsized engines²⁶. The high boosting pressures at the intake manifold of the engine requires increased pre-turbine pressures for creating a positive difference between the exhaust and intake side and allowing short route exhaust gas recirculation. However, as expected, high pre-turbine pressures increase the pumping losses and deteriorate the fuel consumption of the engine.

Figure 5 shows the comparison of the achieved EGR rates (target of 10%) between the baseline engine with LR EGR and the engine with SR EGR system. It is clear that the

baseline engine can achieve maximum EGR rates at full load for most of the engine speeds except very low speeds with EGR values over 9%. On the other hand, the SR system can provide up to a maximum 8% of EGR at top speeds while no exhaust gas recirculates at speeds below 1,500rpm. This clearly means that in the case of an SR system is applied, it is necessary to increase the pressure at the EGR feeding point which as described earlier and found in the literature will deteriorate the fuel economy of the engine.

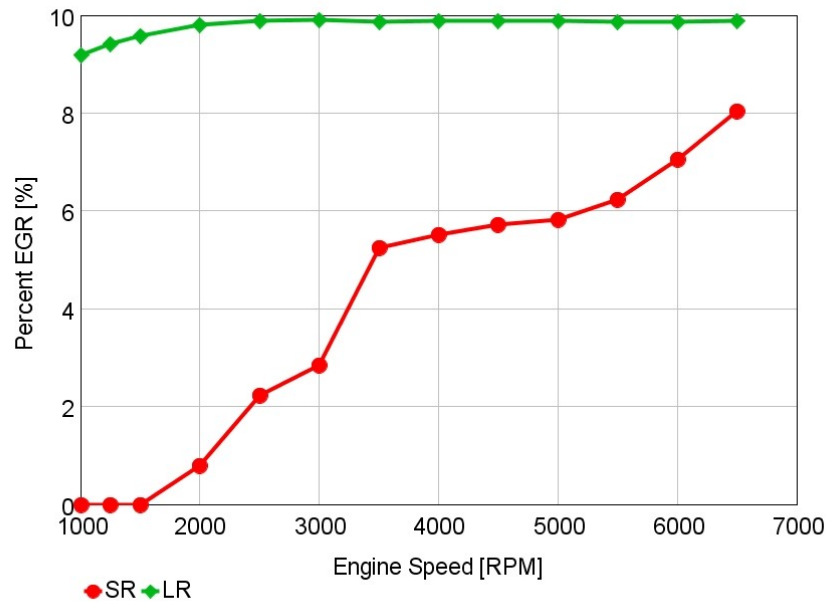


Figure 5. Comparison of the available EGR rates between LR and SR EGR systems out of a targeted 10% rate at full load conditions.

Turbocharger scaling

The implementation of an SR EGR system entails that the exhaust gas collection point is at the exhaust manifold which reduces the mass flow of the gas passing through the turbine. The reduction in energy availability at the turbine inlet results in a reduced turbocharger speed and hence lower mass flow through the compressor. However, as the EGR delivery point is at the intake manifold, the flow requirement through the compressor is also reduced. A scaling exercise has been performed to analyze the effects of a smaller turbocharger on the engine performance and the maximum EGR rates obtained at full load conditions.

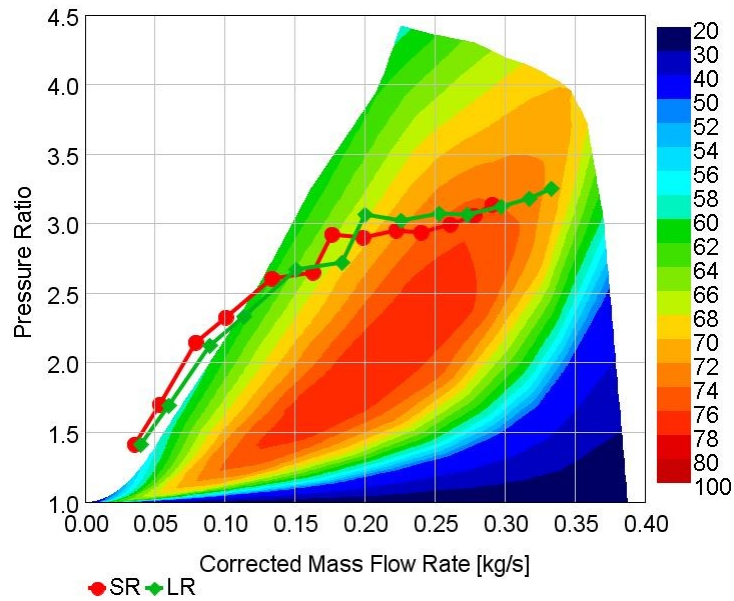


Figure 6. LP compressor isentropic efficiency map (total to total pressure ratio vs. corrected mass flow) for LR and SR EGR models at full load conditions.

As shown in Figure 6, the mass flow passing through the compressor is more than 10% reduced for the engine fitted with an SR EGR system at top engine speeds. Smaller reduction occurs for lower engine speeds due to the reduced levels of obtained EGR. It is evident from the Figure that a 10% scaling would be a realistic approach for the engine fitted with the SR system. However, in this section, scaling of up to 50% smaller than the baseline components has been performed for understanding the effects on engine performance and EGR availability. It should be mentioned that the operation in the surge regions for the low engine speeds is due to the lower load of the supercharger compared to the experimental testing which leads to an increase of the pressure ratio of the first-stage compressor.

It is presented in Figure 7 that a 10% scaling of the turbine can increase the EGR rates for medium to high engine speeds up to a maximum 10% achieved at 6,500rpm. However, the smaller turbine did not have any effects at very low engine speeds with the pre-turbine pressure generated to be considerably lower than the higher speeds. The results show that even a 50% smaller turbine would not provide the targeted EGR rates at low engine speeds.

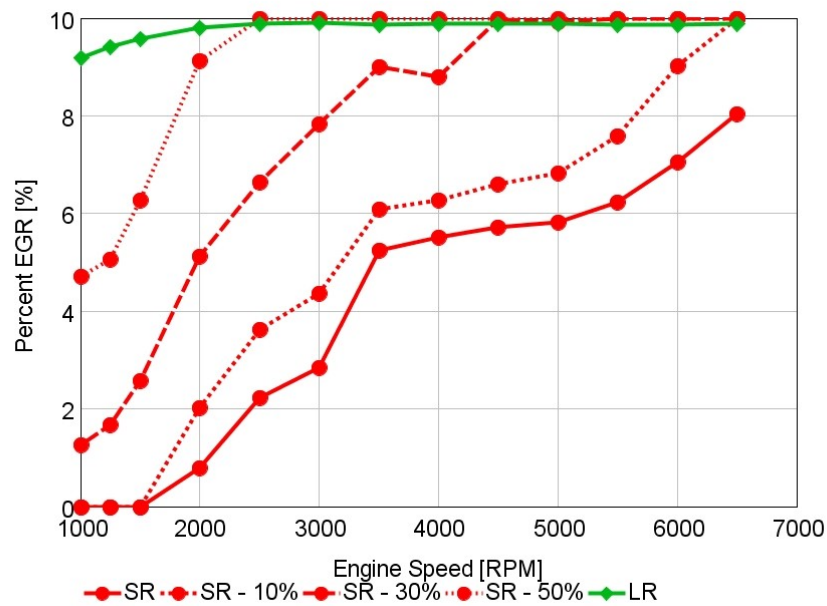


Figure 7. Comparison of the actual EGR rates between LR and SR EGR systems with scaled turbines out of a targeted 10% rate at full load conditions (i.e. 10% indicates a 10% scaling).

Figure 8 illustrates the effects of the turbine scaling on the pre-turbine pressure. As it can be seen, when a 10% scaled turbine is selected the pre-turbine pressure is still at the same levels or even lower compared to the baseline engine due to the decreased mass flow (as a result of the SR EGR). This can ensure that the SR system will not give any fuel penalties, although the targeted EGR rate is still not achievable.

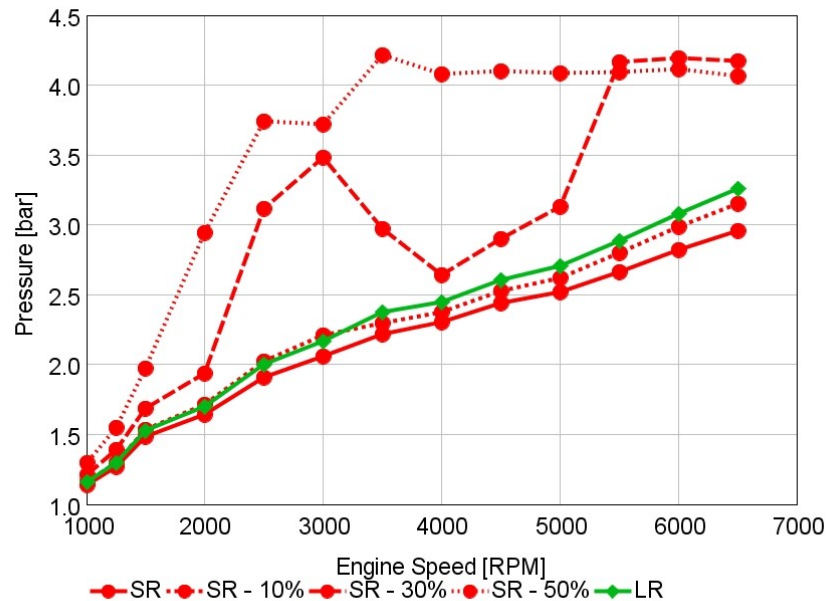


Figure 8. Comparison of actual pre-turbine pressures between LR and SR EGR systems with scaled turbines at full load conditions (i.e. 10% indicates a 10% scaling).

Moreover, it is clear in Figure 9 that a 10% scaling of the turbine and compressor will not have any effects on the performance of the engine. The baseline engine's maximum BMEP can be achieved for all engine speeds compared to the 30% and 50% scaling cases where there is a BMEP deficiency at high engine speeds due to the extreme pre-turbine pressures. It should be mentioned that the BMEP target of the model is controlled by the wastegate position and the speed of the supercharger. An improved control strategy could reduce the very high pre-turbine pressures and BMEP levels observed, especially at low engine speeds. However, at higher engine speeds, increased pre-turbine pressures are unavoidable with the particular engine design which as expected can lead to low engine performance and tendency to knock. On the other hand, the similar levels of pre-turbine pressure and BMEP obtained for the 10% scaled case compared to the LR case, are less likely to trigger engine knocking.

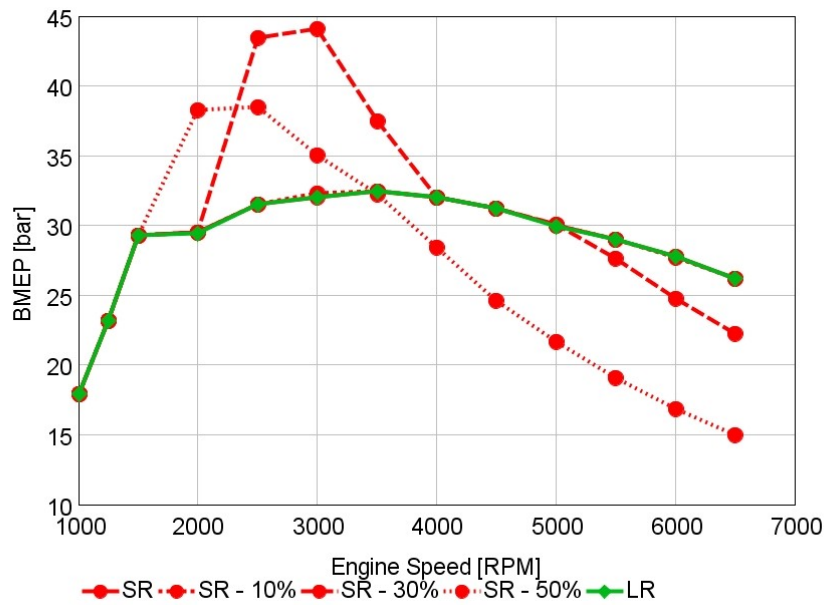


Figure 9. Comparison of the actual BMEP between LR and SR EGR systems with scaled turbines at full load conditions (i.e. 10% indicates a 10% scaling).

Mid-route EGR routing

It can be summarized from the previously presented results that achieving similar SR EGR rates to an LR system can only be achieved with increased pre-turbine pressures, particularly at low engine speeds area. Two-stage boosted engines benefit from a high-pressure compression system, in this case a supercharger, which is used to improve the boosting capacity of the engine and improve transient performance. The activation of the supercharger creates a pressure difference at the intake side prior and post the second-stage compressor which can be taken advantage for introducing higher rates of EGR. This EGR system taking exhaust gases prior the turbine and delivering prior the supercharger is defined in this work as a Mid-Route (MR) EGR system.

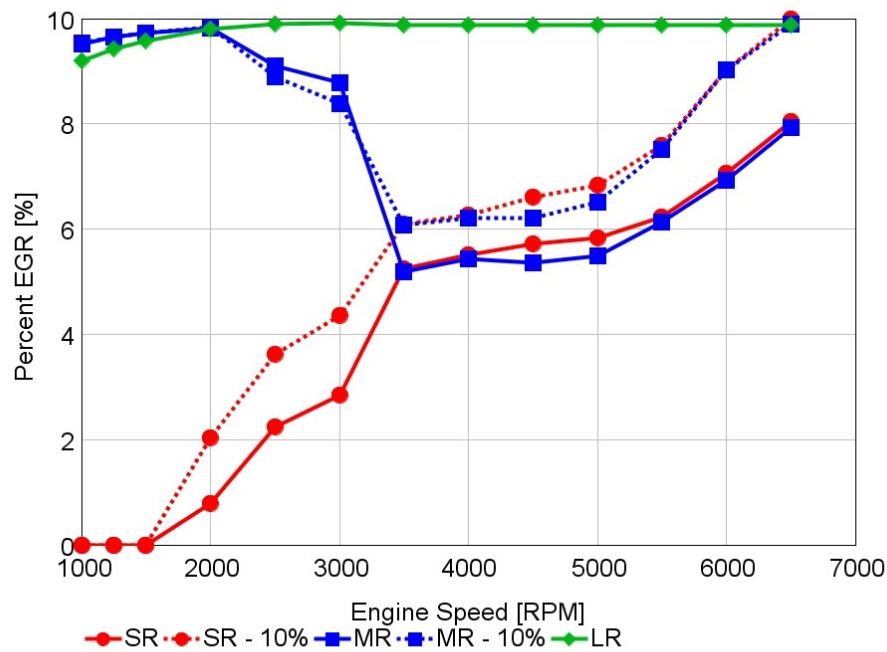


Figure 10. Comparison of the actual EGR rates among LR, SR and MR EGR systems out of a targeted 10% rate at full load conditions (i.e. SR - 10% indicates a 10% scaling of turbine and compressor).

Figure 10 demonstrates that the newly introduced Mid-route EGR system provides similar EGR rates to the SR system for medium to high engine speeds. On the other hand, for engine speeds of 3,000rpm and below, the MR system can provide maximum EGR rates of more than 8% due to the positive pressure difference between the exhaust and the intake prior the supercharger after its activation.

Hybrid EGR systems

Although the MR EGR system increases the maximum EGR rates at low engine speeds, the rates achieved are still below the targeted EGR values that need to be applied in the specific engine. It seems that none of the SR or MR can obtain maximum EGR rates unless increased pre-turbine pressures are applied which consequently may deteriorate the fuel economy of the engine. Therefore, it is mandatory for the short or mid-route systems to be assisted by a long route EGR path which will work as a cover up in a hybrid EGR system.

Figure 11 shows that the hybrid EGR systems (SR+LR or MR+LR) can achieve the targeted 10% EGR rate for all engine speed with the LR system assisting only when necessary with the minimum amount needed.

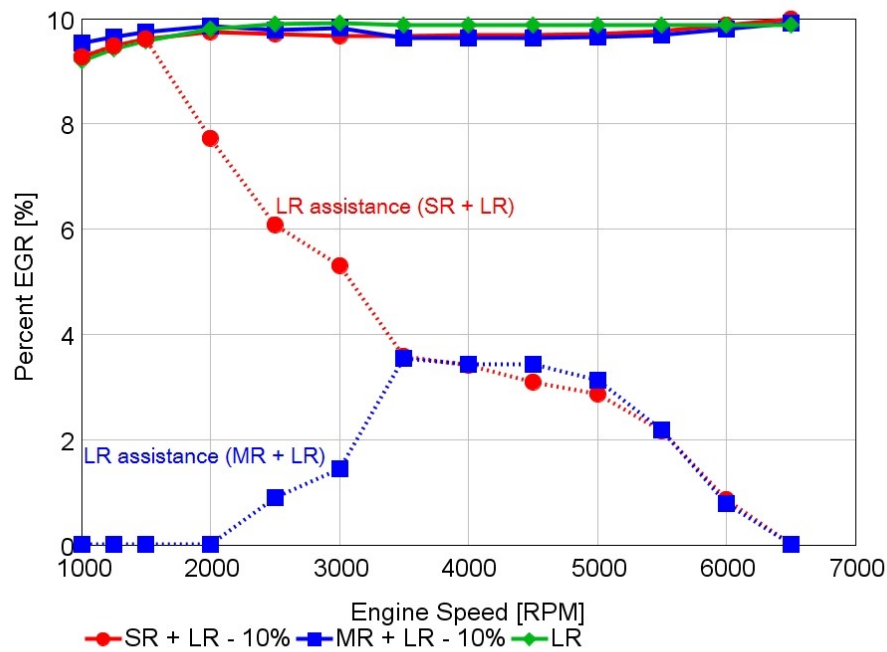


Figure 11. Maximum obtained EGR percentage comparison between hybrid (10% scaling) and baseline cases at full load conditions (i.e. SR+LR - 10% indicates a 10% scaling of turbine and compressor).

It is illustrated in Figure 12 that the pre-turbine pressure (mainly responsible for increased pumping losses and fuel economy deterioration) for most of the hybrid systems does not exceed the pressure of the baseline engine. The only exception is the SR+LR hybrid with a 10% scaled turbine at low engine speeds because no SR EGR is obtained under these operating conditions. Moreover, it is evident that the MR+LR hybrid system with scaled components can provide a decreased pre-turbine pressure which can potentially lead to reduced pumping losses and improved fuel economy.

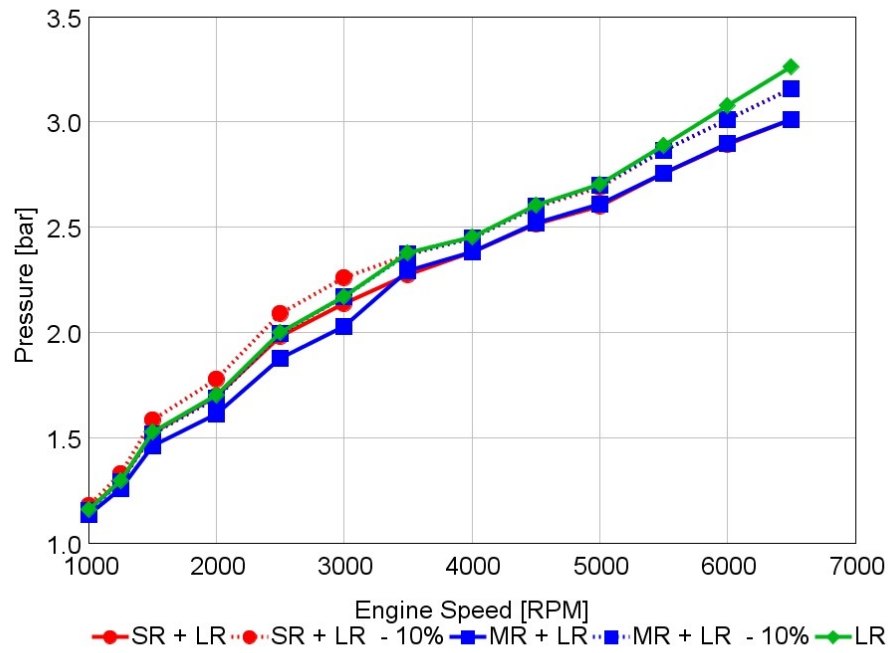


Figure 12. Pre-turbine pressure comparison between hybrid and baseline cases at full load conditions (i.e. SR+LR - 10% indicates a 10% scaling of turbine and compressor).

Figure 13 shows the pumping mean effective pressure comparison between the baseline and hybrid cases. As can be seen from the Figure, the PMEP of the MR and SR hybrid cases with the original turbocharger exhibit increased PMEP levels at low engine speeds due to the incapacity of the compressor to reach the boosted pressure of the baseline engine after part of the exhaust gases have been diverted prior the turbine. However, for the cases with the scaled turbocharger, the PMEP values are of the same level as in the baseline engine. At higher engine speeds, the hybrid systems provide lower values of negative PMEP compared to the baseline engine which translates to lower values of positive work. This is happening although models with the hybrid systems exhibit considerably lower pre-turbine pressures. This leads to the conclusion that for high engine speeds, there is room for further reduction of the turbine size selected when short or mid-route EGR is applied.

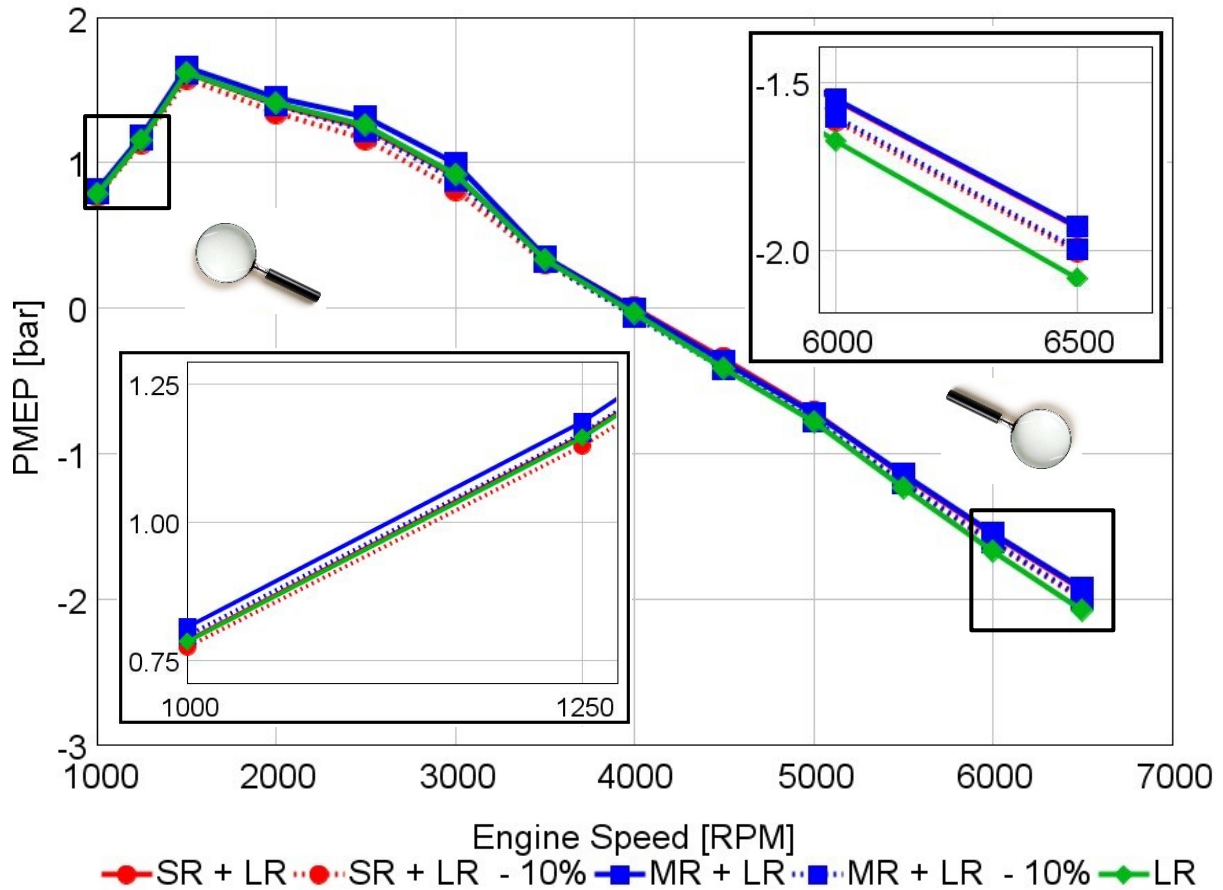


Figure 13. PMEP comparison between hybrid and baseline cases at full load conditions (i.e. SR+LR - 10% indicates a 10% scaling of turbine and compressor).

However, the introduction part of the exhaust gases prior the supercharger (MR hybrid system) increases the mass flow through the second boosting device compared to an SR hybrid system. Figure 14 illustrates the power consumption of the supercharger for the three different EGR systems. It is clear that with the SR hybrid system, the lowest amount of power is consumed by the SC due to the reduced mass flow as well as the increased post-compressor resulted by the smaller turbocharger and no SR EGR flow.

The engine with the MR hybrid system provides a slight increase in the supercharger's power consumption at low engine speeds. This increase is due to the reduced boosted pressure achieved by the first compressor as for this case the MR path obtains full EGR rates. At medium engine speeds, where the maximum MR EGR obtained drops and with the help of a smaller turbine, the pressure prior the supercharger is increased and the power consumption of the supercharger is reduced.

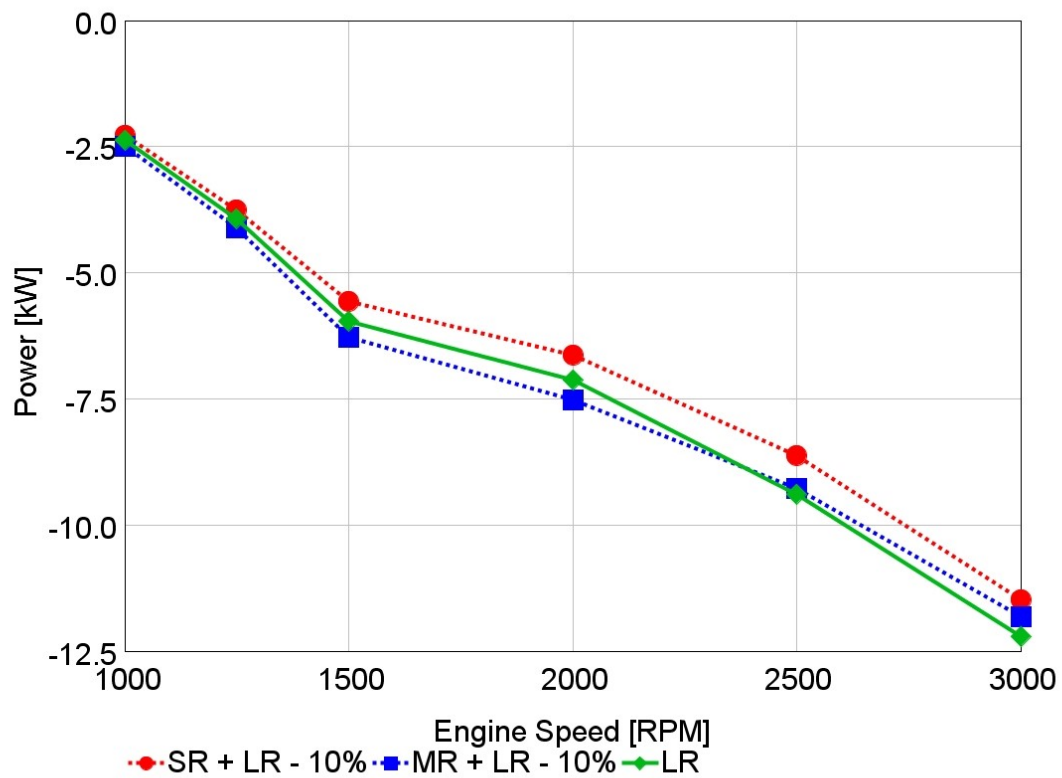


Figure 14. Comparison of the average power consumption by the supercharger between the baseline and scaled hybrid cases at full load conditions.

Finally, Figure 15 confirms that the increased power consumption at low engine speeds, as a result of the MR EGR implementation, deteriorates the fuel economy of the engine. Results reveal that the engine shows a penalty of up to 1.4% compared to the LR system and 2.6% in relation to the SR hybrid system in fuel consumption at full load and engine speeds below 2,500rpm. The penalty is lower for medium engine speeds where a lower amount of MR exhaust gas recirculation is obtained. No BSFC penalty occurs at high engine speeds due to deactivation of the supercharger.

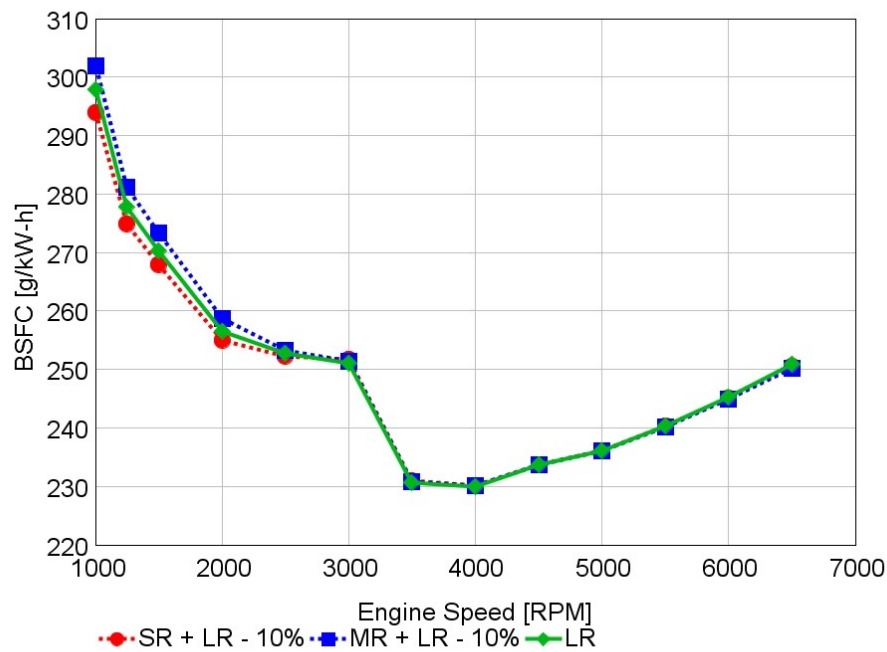


Figure 15. Comparison of the average break specific fuel consumption between the baseline and scaled hybrid cases at full load conditions.

Transient Analysis

The transient analysis has been performed to evaluate the benefits and drawbacks of the MR hybrid EGR system over the LR and SR hybrid regarding EGR filling and purging time as well as load response time. For all the studies, the turbine and compressor for the hybrid EGR cases are scaled by 10% compared to the LR EGR case. **The smaller turbocharger could potentially result in lower inertia forces and faster response time, however, for the purpose of this study the inertia change of the scaled turbocharger has not been considered.**

Load tip-in event

The load tip-in transient from 5bar BMEP to full load was performed for the baseline and hybrid EGR models with 10% scaled turbochargers for engine speeds of 1,500rpm and 3,500rpm. For the purpose of achieving a faster load increase, the EGR valve was shut and the time required for the initial 10% EGR rate to purge is shown in Figure 16. As it can be seen in the Figure, for the 3,500rpm engine speed, the LR EGR system shows the longest purging delay of EGR compared to the hybrid cases that exhibit an almost rapid response.

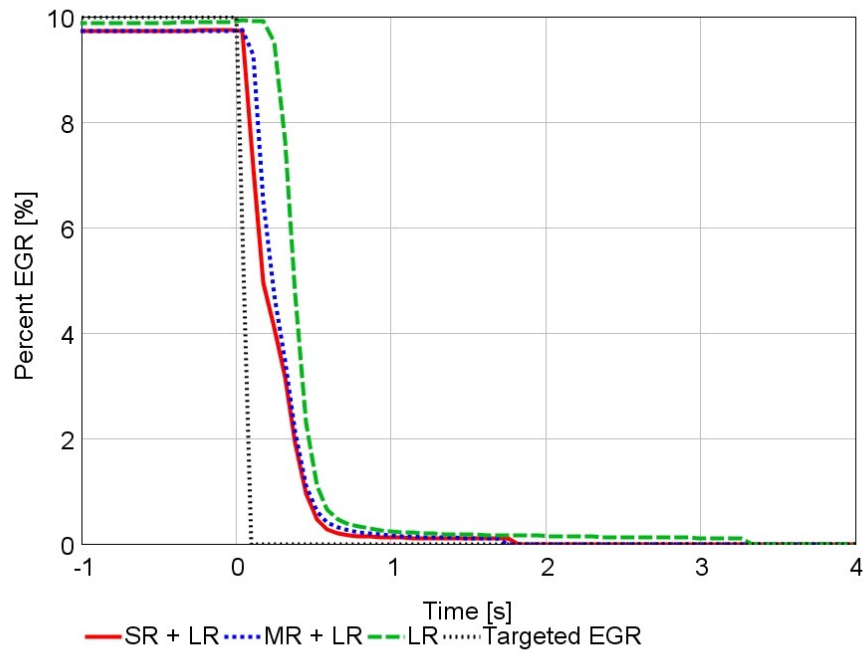


Figure 16. EGR flow purging time for various routes at 3,500rpm, full load conditions.

The SR hybrid EGR system provides the fastest purging time due to the very short piping length (1.4m) between the EGR valve and the combustion cylinders. For the MR hybrid case, although the length of the pipes is slightly smaller (2.4m) than the LR system (2.6m), the purging time is improved considerably up to 75% (at the point where EGR rate starts to descend). This is due to the increased post-compressor pressure after the load tip-in which works as a secondary valve at the EGR supply point and traps any post EGR valve gases between the EGR valve and the supply point. This does not allow the post EGR valve gases to purge into the combustion cylinder compared to the LR system where all the exhaust gases passing the EGR valve are delivered to the cylinders. The rapid purging of the EGR flow for the hybrid system combined with the smaller turbine applied provides a reduction in the time required to achieve full load by more than 25% as shown in Figure 17.

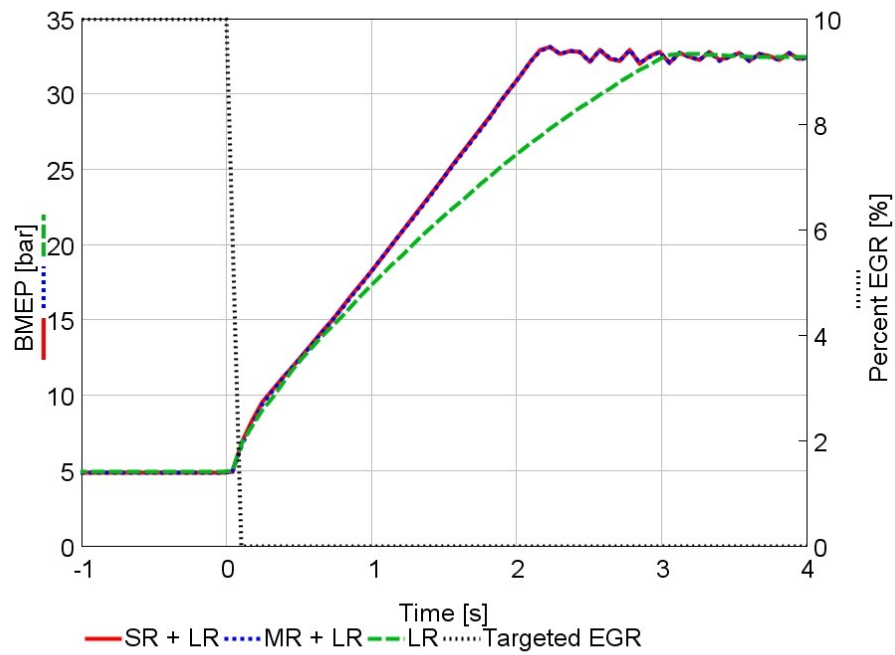


Figure 17. Load tip-in results for different EGR routes at 3,500rpm.

The EGR purging time for the low engine speed follows a similar trend as shown in Figure 18. It is clear that the MR hybrid case provides the fastest purge compared to the LR baseline case and the SR hybrid which at this engine speed consists of only long route EGR as SR cannot be obtained at low engine speeds (due to the negative pressure difference between the exhaust and intake sides).

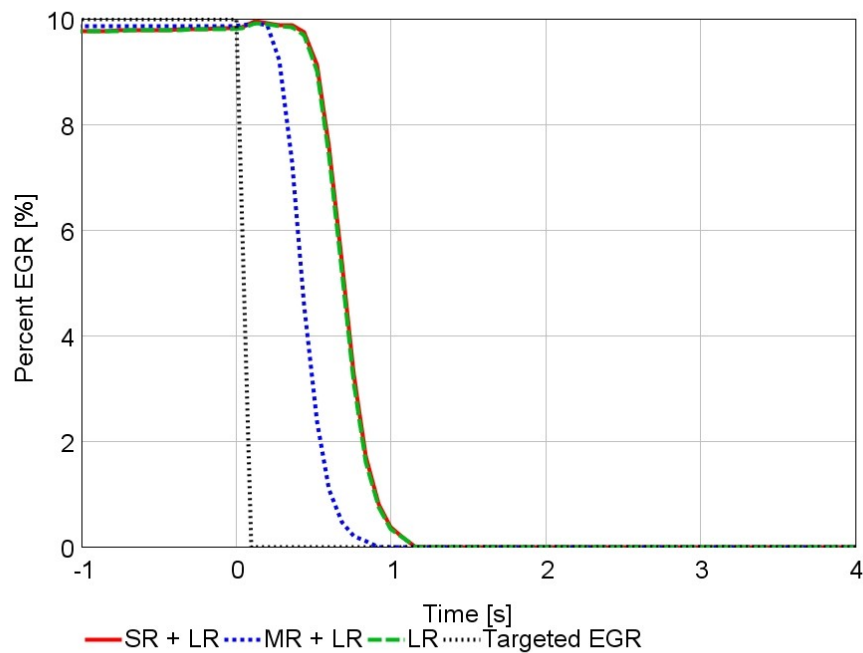


Figure 18. EGR flow purging time for various routes at 1,500rpm, full load conditions.

However, Figure 19 shows that the benefit of the transient response time is smaller compared to the 3,500rpm case. The response time for the two hybrid cases is the same which provides evidence that the time improvement is mainly due to the smaller turbine with lower inertia applied compared to the baseline engine and not the faster EGR purging time.

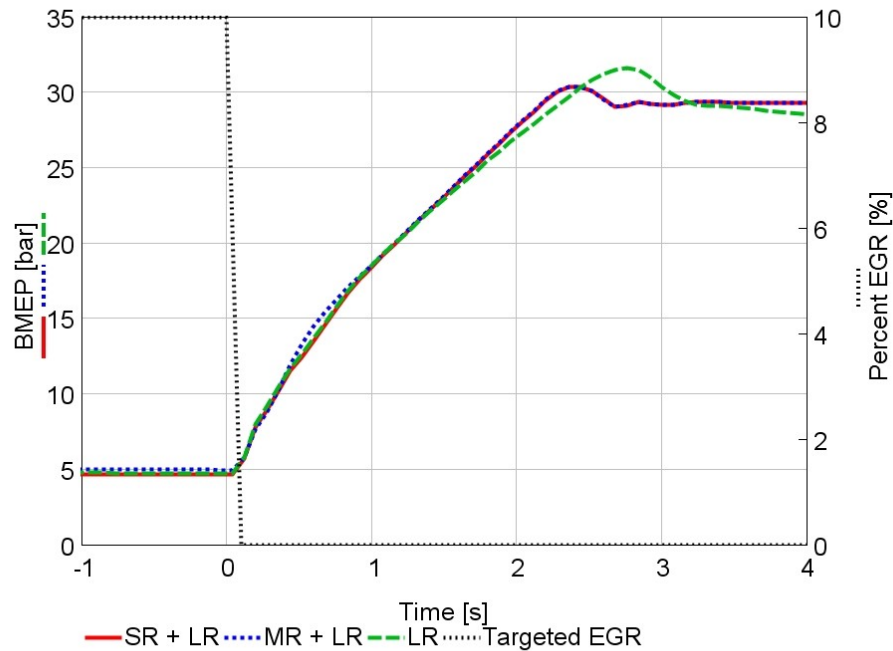


Figure 19. Load tip-in results for different EGR routes at 1,500rpm.

EGR ‘tip-in’ event (full load)

The second part of the transient investigation provides the results of an EGR ‘tip-in’ event at full load conditions. Figure 20 outlines that the SR hybrid system provides the fastest EGR increase for the 3,500rpm case due to the short distance the exhaust gases have to travel.

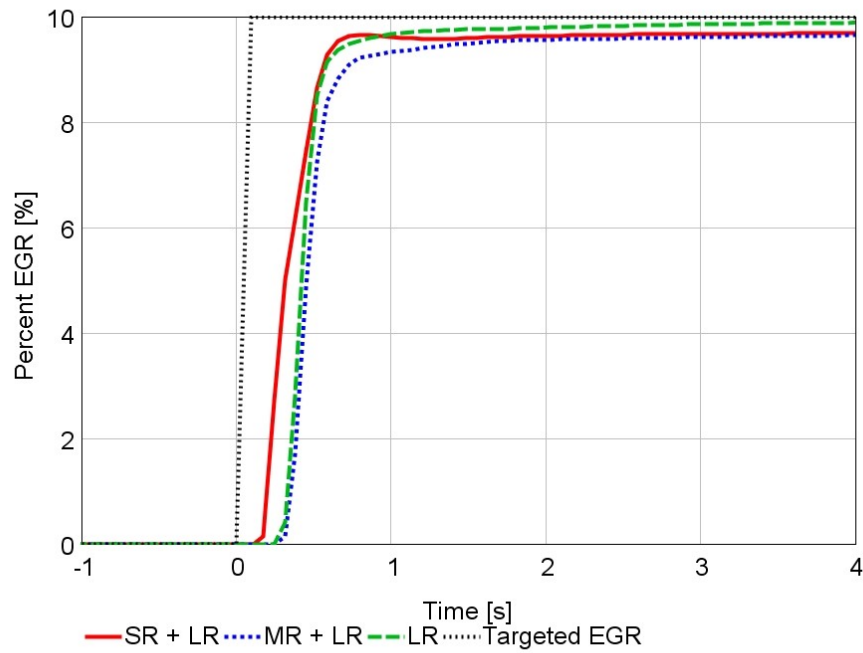


Figure 20. EGR filling during a transient event for various routes at 3,500rpm, full load conditions.

It is remarkable though that despite the distance needs to be traveled by the MR EGR flow is shorter than the LR flow, the MR hybrid system provides a slower response. This is happening because the EGR supply point for the MR hybrid system is prior the turbine which causes a reduction in the mass flow passing through the turbine and hence reduces the boosting performance of the compressor instantaneously (Figure 21), resulting in a negative pressure difference between the exhaust and intake sides. This BMEP fall shown in Figure 22 delays the delivery of the gases from the exhaust manifold to the engine's cylinders. A similar phenomenon is also applied to the SR hybrid system. However, it is backed by the very small distance between the collection and supply points. The BMEP drop occurring for the LR case does not affect the EGR supply because the pressure difference at the low-pressure collection and supply points remains positive.

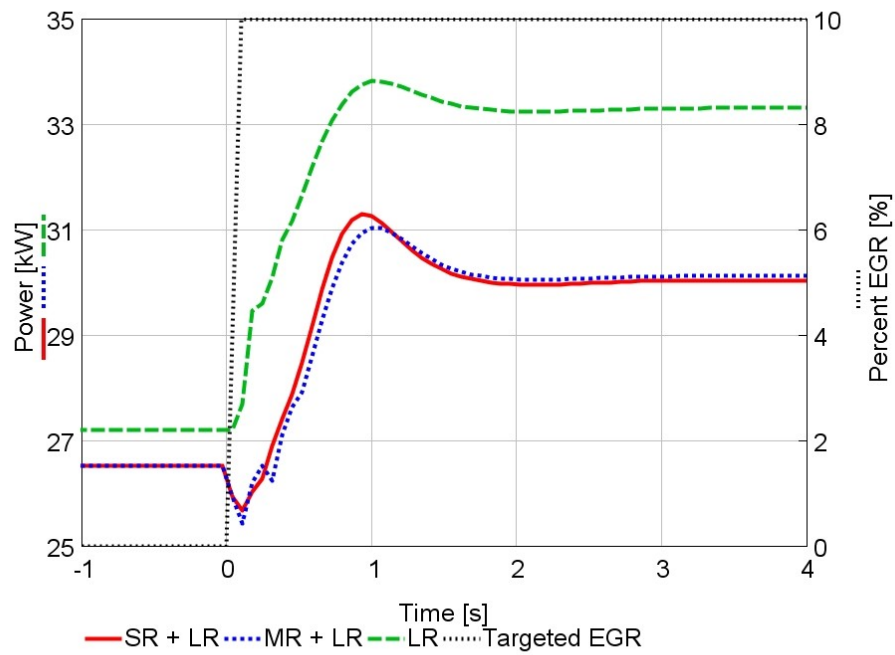


Figure 21. **LP compressor's output power** during an EGR tip-in event for various EGR routes at 3,500rpm, full load conditions.

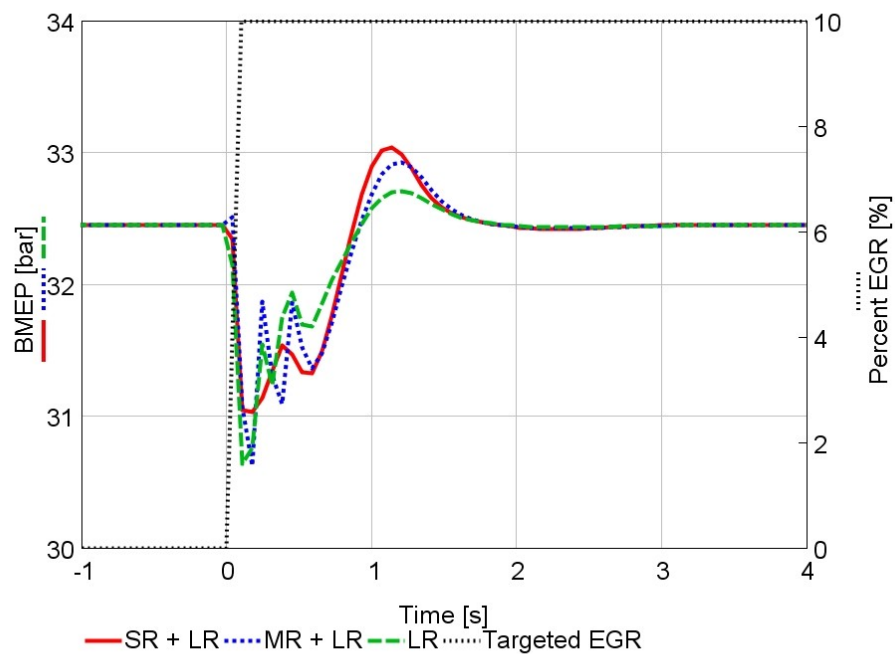


Figure 22. BMEP fluctuations during an EGR tip-in event for various EGR routes at 3,500rpm, full load conditions.

In Figure 23 the results of a 10% EGR ‘tip-in’ event for an engine speed of 1,500rpm are presented. In this case, the MR hybrid case provides the fastest response compared to the baseline and SR hybrid system which consists of LR EGR only at this engine speed.

The difference on the EGR response time for this engine speed is a result of the supercharger which is activated and pressurizes the intake flow during the EGR supply event creating a positive pressure difference between the EGR collection and supply (prior the supercharger) points.

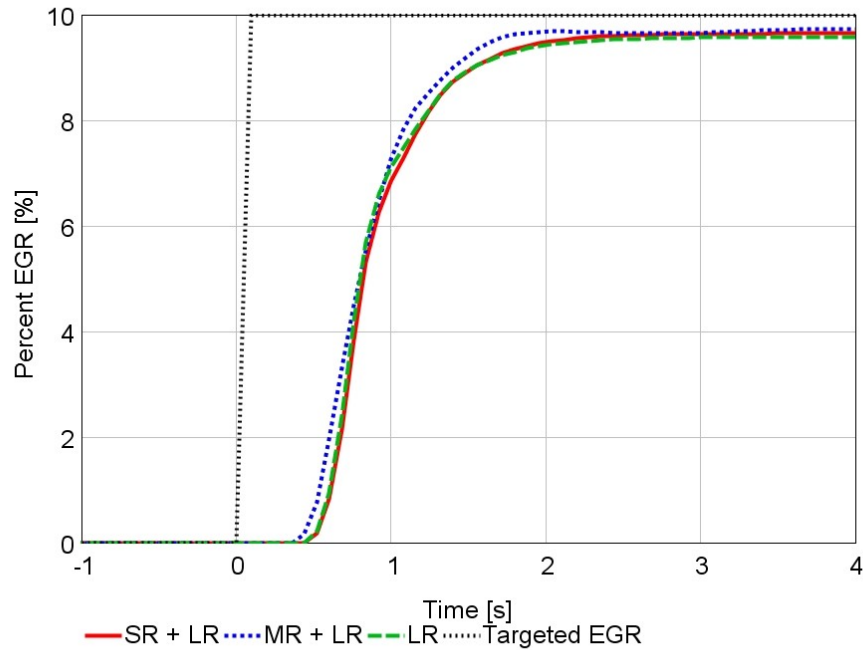


Figure 23. EGR filling during transient for various routes at 1,500rpm, full load conditions.

Coolers sizing

The sizing of the EGR coolers and the intercooler for the three different EGR supply systems with LR EGR, SR and MR hybrids was performed in this section. The cooling performance of the aftercooler placed right after the second boosting device was not modified and it was kept constant at the baseline engine's values for all the cases tested.

LR EGR cooler sizing

The implementation of a hybrid EGR system indicates that there is the likelihood of an additional EGR cooler needed. However, the smaller amount of gases passing through the LR EGR cooler allows the implementation of an LR EGR cooler with smaller cooling capacity.

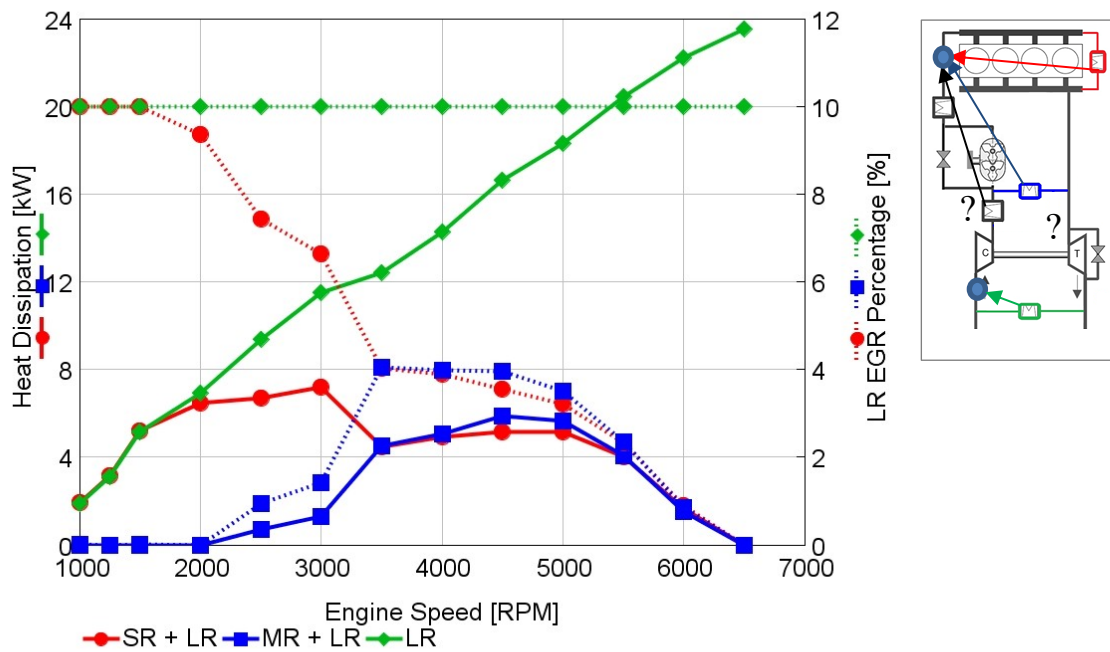


Figure 24. LR EGR cooler heat dissipation vs. obtained LR EGR rates for various systems at full load conditions (schematic shows cooler under investigation (?), cooler not in use (X) and arrows indicate the temperature control target of each cooler).

Figure 24 illustrates that for the baseline engine, the highest heat dissipation of 24kW occurred at top speeds due to the increased mass flow and increased EGR cooler's inlet temperatures. For the MR hybrid case, the maximum cooling capacity is reduced by 75% or more and occurs at medium engine speeds where the highest LR EGR assistance occurs. On the other hand, for the SR hybrid, the maximum heat dissipation occurs at 3,000rpm and is roughly 65% smaller than the baseline engine. In overall, **the maximum heat needs to be dissipated** from the LR EGR cooler of the MR or SR hybrid and LR systems is around 6, 7.5 and 24kW respectively and occurs at different engine speeds.

Intercooler sizing without LR EGR cooler

Considering the previous Figure, the implementation of a hybrid EGR system reduces the amount of LR EGR, particularly for the MR hybrid case. Therefore, this section analyses whether an LR EGR cooler is necessary for the operation of the hybrid EGR systems. Eliminating the necessity of an extra cooler is likely to reduce the production cost of the vehicle as well as save weight and space which is a critical issue for modern engines.

Figure 25 shows that the MR hybrid case without an LR EGR cooler exhibits compressor inlet temperatures lower than the baseline engine for low and high engine speeds where there is a minimum amount of LR EGR obtained. For medium engine speeds where LR EGR assistance is provided, the temperature increases by more than 10 degrees (325K) at 3,500rpm. However, this is a considerably low increase (occurring only at full loads) which is not expected to have any considerable effects on the compression process and is not likely to cause a violation of the compressor's wheel maximum temperature limit. The implementation of an EGR mixer would also be advisable to improve the air-EGR homogeneity and avoid hot spot areas near the compressor²⁷.

On the other hand, the SR hybrid system exhibits high pre-compressor temperatures up to 360K at low engine speeds where no SR EGR is obtained. This high temperature would result in disproportionately high compression final temperatures, increased losses and a higher work of compression. Also, the very high inlet temperature for the SR hybrid case is likely to violate the compressor's wheel temperature maximum limit.

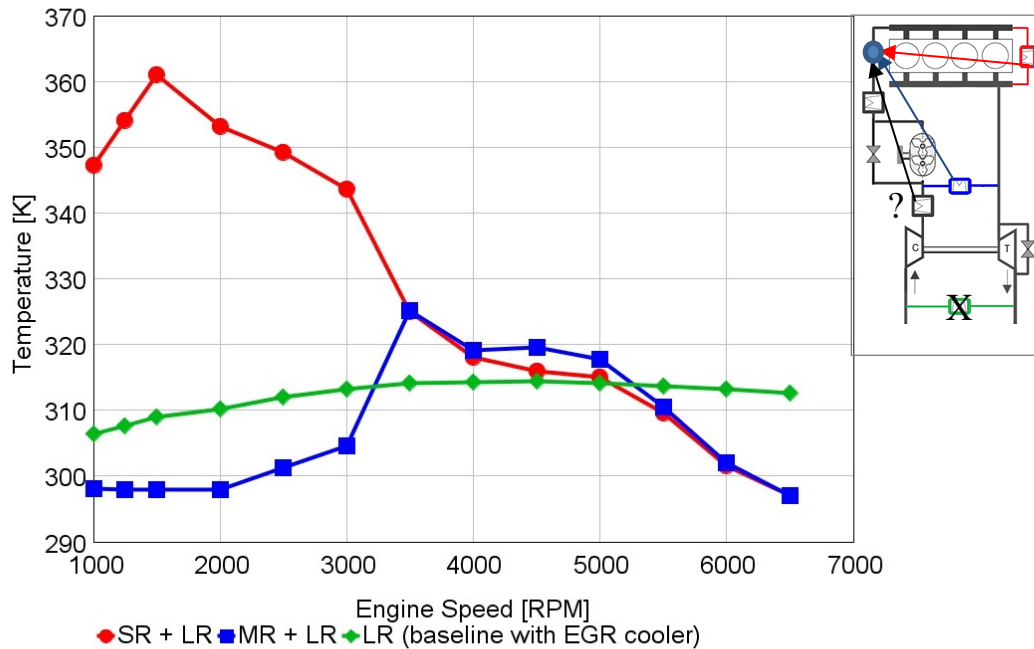


Figure 25. Compressor inlet temperature for cases without LR EGR cooler at full load conditions (schematic shows cooler under investigation (?) and arrows indicate the temperature control target of each cooler).

Moreover, the increased compressor's outlet temperatures for the SR hybrid requires a larger amount of heat dissipation by the intercooler which as shown in Figure 26 is up two times higher at low to medium engine speeds. On the other hand, the slight temperature increase at medium engine speeds for the MR hybrid does not require a larger intercooler. Also, the heat dissipation needed for the top speed is almost 30% lower (-14kW when assuming a fully efficient MR EGR cooler) compared to the baseline engine due to the reduced temperature and mass flow rate.

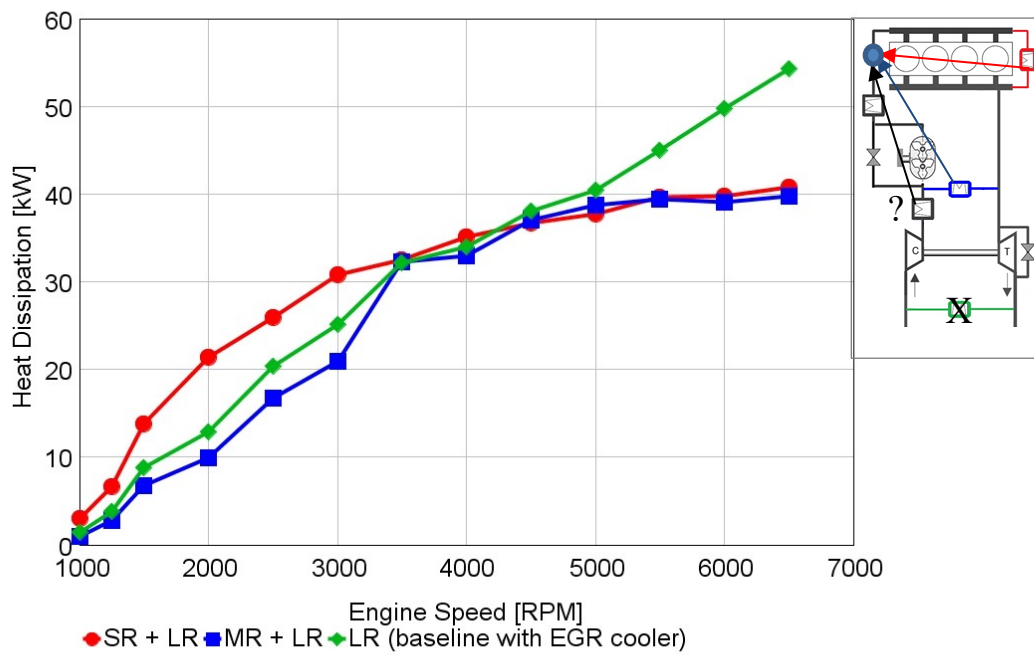


Figure 26. Comparison of the heat dissipation out of the intercooler between cases without LR EGR cooler and the intercooler of the baseline engine at full load conditions (schematic shows cooler under investigation (?) and arrows indicate the temperature control target of each cooler).

IC sizing with various MR or SR EGR cooler sizes

The implementation of an additional short or mid-route EGR system would require the resizing of the intercooler based on the cooling capacity of the additional EGR cooler. Figure 27 presents a detailed analysis of the IC's maximum heat dissipation occurring at high engine speed for various sizes of MR and SR EGR coolers. The study includes EGR coolers of similar capacity to the LR EGR cooler of the baseline engine and up to 50% increased cooling capacity. The results are presented in kW/kg/min due to the different amount of mass flow rates passing through the EGR coolers that affect their performance.

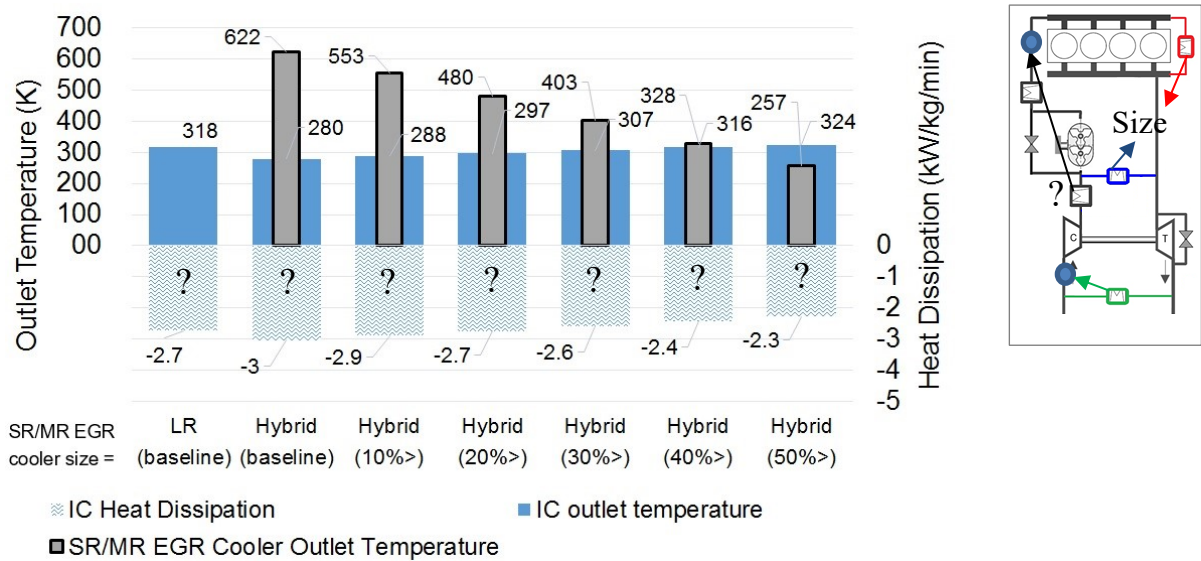


Figure 27. Intercooler's heat dissipation demands and outlet temperatures for different sizes of SR or MR EGR coolers at full load conditions. (values in the Figure are the mean of SR and MR hybrid systems, results span in the range of $\pm 0.6\%$ for the IC's outlet temperature and $\pm 4\%$ for the SR/MR EGR outlet temperature)

As shown in Figure 27 the implementation of a cooler with the same capacity of the LR EGR cooler in the baseline engine would require extensive **heat dissipation** from the IC to reach a steady temperature of 318K at the intake manifold. The post-intercooler temperature would need to be dropped down to 280K which is not practical and achievable with the cooling systems fitted in a vehicle. That means that the SR or MR EGR cooler needs to be of a greater capacity than the baseline LR EGR cooler. The results reveal that with an SR or MR EGR cooler 20% to 30% larger, the IC's capacity per kg/min of flow remains the same or slightly decreases (reduced up to 15% on the total energy needs to be removed due to a reduced mass flow). At the same time the post-IC temperatures are in a reasonable range. The MR or SR EGR cooler's capacity is in the range of a maximum 17.5 kW/kg/min of heat dissipation which can reach a total of 32kW of heat removed for maximum EGR flows at high speeds.

IC sizing for different MR options

The MR hybrid EGR system collects exhaust gases upstream the turbine and delivers post the compressor of the engine with the EGR cooler used for removing the unnecessary heat. The exhaust gases can be potentially delivered prior the IC making the implementation of an EGR cooler optional.

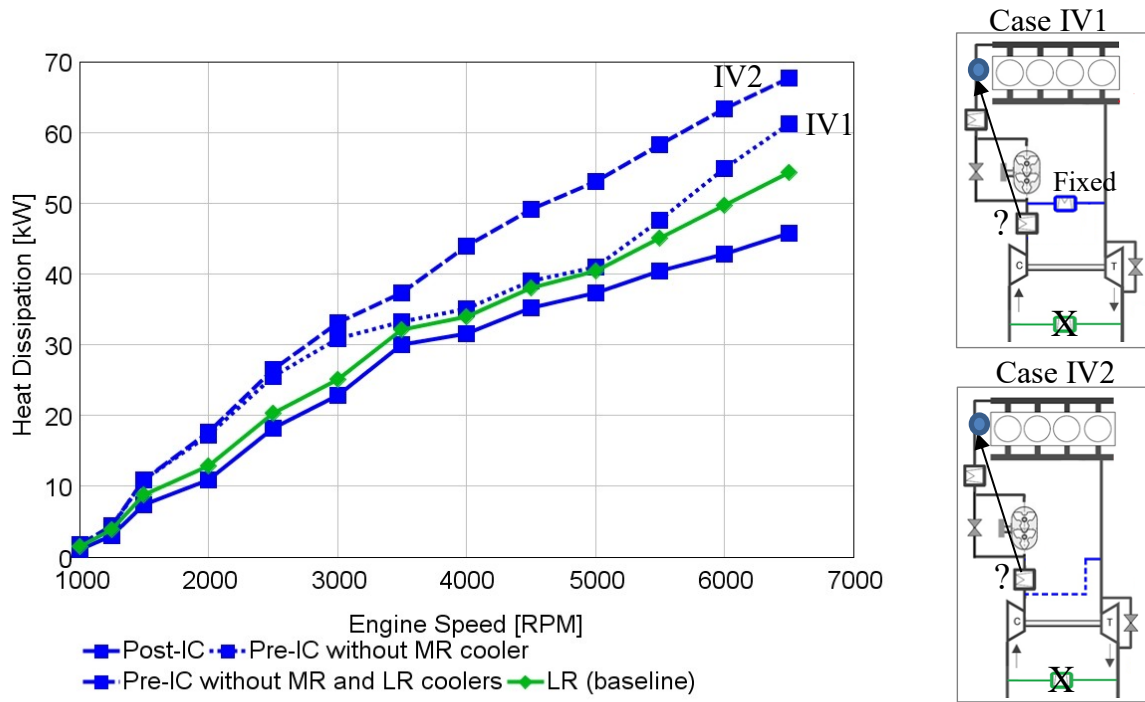


Figure 28. Comparison of the heat dissipation out of the intercooler between different MR hybrid options and the baseline model at full load conditions (a 20% larger MR EGR cooler than baseline is applied for the Post-IC case).

Figure 28 shows that for the case with the MR EGR delivered post the IC, the heat removed by the intercooler is up to 15% smaller than the baseline engine due to the elimination of hot gases coming from the LR EGR system. The implementation of a pre-IC MR system without an MR EGR cooler increases the heat that needs to be dissipated by the IC at low and high engine speeds where high MR EGR rates are achieved. The increase is up to 7kW high (>10%) for top engine speeds while this goes up to 15kW (>25%) when no LR EGR cooler is implemented for cooling down the LR EGR flow.

Conclusions

This paper has investigated the benefits and drawbacks of a mid-route EGR system for a heavily downsized two-stage boosted engine. From the research that has been conducted, it is possible to conclude that the mid-route EGR route is a feasible alternative to the short route system favouring of reduced pre-turbine pressures whilst obtaining high EGR rates.

The investigation revealed that the MR system could achieve maximum EGR rates at low engine speeds where the supercharger is activated, unlike a SR path. However, for achieving maximum rates of EGR across all engine speeds, a hybrid system of SR or MR with LR EGR is required. The study showed that the MR+LR hybrid system reduces the pre-turbine pressure compared to SR+LR, but increases the flow and power consumption of the supercharger. There was a noticeable increase in BSFC for the MR EGR hybrid system (1.4%); this was due to increased supercharger work despite a reduction in pumping losses. With the SR hybrid EGR, the BSFC was reduced by 2.6% through improvements in pumping work.

The transient analysis revealed that the MR hybrid system benefits of up to 50% reduction in the EGR purging time, similar to the SR hybrid system, during load tip-in events. Also, the faster EGR purging combined with a smaller turbine implemented improved the load transient response time by up to 25% compared to the baseline engine. On the other hand, the

MR hybrid system demonstrated a slight delay compared to the LR and SR hybrid systems during an EGR ‘tip-in’ event due to the instantaneous drop of the compressor’s output power as a result of the pre-turbine diverted flow. However, this is not the case for low engine speeds where the supercharger is activated and the MR hybrid system provides the fastest response compared to the LR and the SR hybrid that requires a high amount of LR EGR assistance.

A cooling capacity study indicated that the MR+LR hybrid system could operate without the need of an LR EGR cooler while maintaining compressor outlet temperatures below 220°C. For this system, an EGR cooler with specific cooling capacity 20% to 30% larger than the LR only EGR cooler is required. At the same time, the intercooler could be 15% smaller (~45kW) than the baseline engine.

If the MR EGR cooler were to be combined with the intercooler, the intercooler needs to provide a cooling capacity of up to 10% higher (~61kW) when an LR EGR cooler is fitted. The heat dissipation demand is increased to over 25% (~68kW) greater than the baseline when the LR EGR cooler is omitted.

On the basis of the promising findings presented in this paper, work on any remaining issues is continuing and will be presented in future papers. Future work will involve simulation studies at low and medium engine loads for accessing the maximum obtained EGR rates and coolers’ performance for the different EGR routes. Finally, the simulation studies can be supported by experimental investigations.

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Declaration of conflicting interests

The authors declare that there is no conflict of interest.

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Abbreviations

AC	Aftercooler
BMEP	Brake mean effective pressure
BSFC	Brake specific fuel consumption
DISI	Direct injection spark ignition
DPF	Diesel particulate filter
EGR	Exhaust gas recirculation
EUDC	Extra-urban driving cycle
GDI	Gasoline direct injection
HC	Hydrocarbons
HP	High pressure
IC	Intercooler
IM	Intake manifold
LP	Low pressure
LR	Long route
MR	Mid-route
NA	Naturally aspirated
NEDC	New European driving cycle
PCCI	Premixed charge compression ignition
PM	Particulate matter
PMEP	Pumping mean effective pressure
rpm	Revolutions per minute
SC	Supercharger
SR	Short route
TC	Turbocharger